AN INVESTIGATION OF PICKING NOISE
IN AN AUTOMATIC LOOM

A THESIS

Presented to

The Faculty of the Division of Graduate

Studies and Research

By

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In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

Georgia Institute of Technology

October, 1974

AN INVESTIGATION OF PICKING NOISE

IN AN AUTOMATIC LOOM

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ACKNOWLEDGMENTS

The author wishes to express his sincere appreciation to all those individuals who have contributed to this work by their assistance and encouragement.

Gratitude is expressed to the School of Textile Engineering for providing a fly shuttle loom for the experiments. Appreciation is extended to Dr. David M. Sanborn and Dr. Amad H. Tayebi for their review of this work and their help during its development. The author would also like to acknowledge the technical assistance of Messrs. Gene Clopton, Howard Dotson, Tony Chao, and Barry Strauss.

Special thanks are extended to Dr. Allan D. Pierce who originally suggested this problem and provided guidance during the research as faculty advisor.

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SUMMARY

Surveys have shown that weaving mills, where sometimes hundreds of fly shuttle looms exist under one roof, are among the noisier industrial environments with noise levels on the order of 102 dB(A) [Karplus and Bonvallet, 1953]. Hanson (1974) estimates that over 250,000 people may be subjected to weave room noise on a daily basis. Such exposure constitutes a hearing damage risk [Burns, Hinchcliffe, and Littler, 1964] and detracts from the quality of life for those affected workers. Noise reducing modifications for existing fly shuttle looms are desirable for this reason.

This work presents an analytical estimation of the octave band sound pressure levels at a reference point due to various vibrating fly shuttle loom components. The analytic models (based on component surface dimensions, octave band normal surface acceleration, and radial distance to a reference point) predict the two picking sticks to be major noise sources in each octave band. In the 500 Hz through 8000 Hz octave bands (where loom noise levels are greatest) the two picking sticks are separated from the third loudest source by 24 dB or more. Two possible methods of picking stick noise control (lagging and surface damping) are suggested and future research goals are discussed.

CHAPTER I

AN INTRODUCTION TO THE FLY SHUTTLE LOOM

The loom, a machine which interlaces two sets of threads (the warp and the weft) at right angles to one another, has been used to weave cloth for thousands of years. Although the basic design dates back to antiquity, several major modifications were made during the period of 1733 to 1894. John Kay's invention of the flying shuttle in 1733 marked the beginning of an era which saw the addition of power to the loom (Edmund Cartwright in 1786) and the total automation of the loom (James Northrop in 1894) [English, 1969]. Today, most natural fiber weaving is done on fly shuttle looms.

A typical fly shuttle loom is shown in Figure 1. For convenience of referral, some close-up photos of various parts of a fly shuttle loom are shown in Figures 2-5. During normal operation of the loom, the weft carrying shuttle is repeatedly thrown back and forth across the width of the loom between alternately raised and lowered warp threads, the space between the threads being termed the shed. The lower threads rest upon a wooden crossmember termed a sley and the shuttle partly slides on this. The acceleration of the shuttle is accomplished by means of

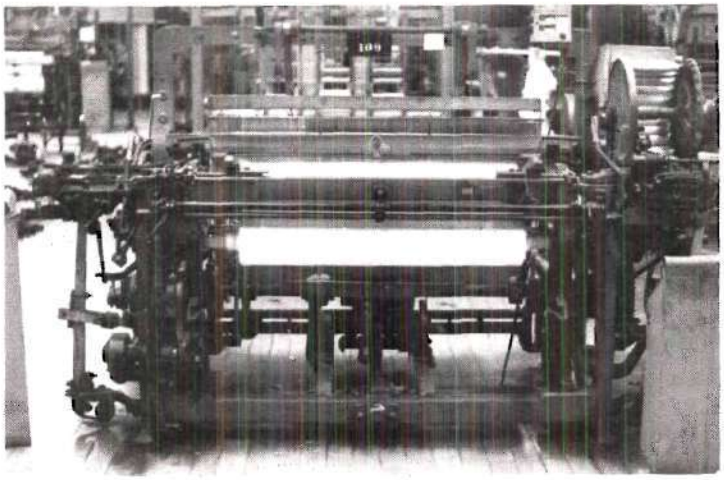


Figure 1. A Typical Fly Shuttle Loom
(Draper 5659)

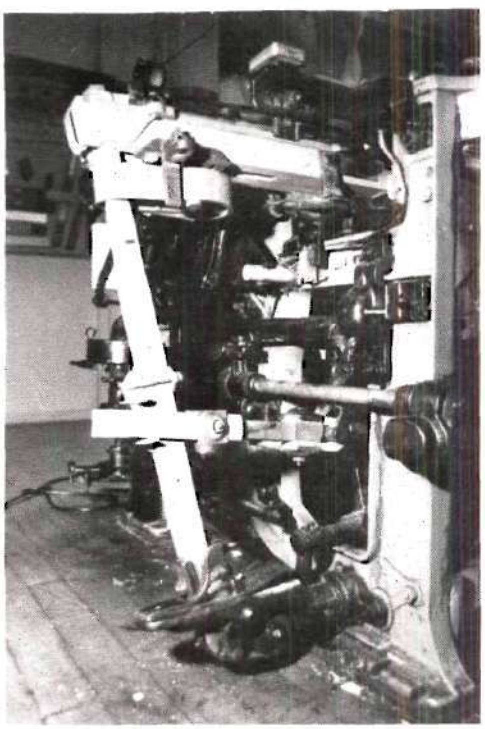


Figure 2. The Left Picking Stick of the Hunt Automatic Loom

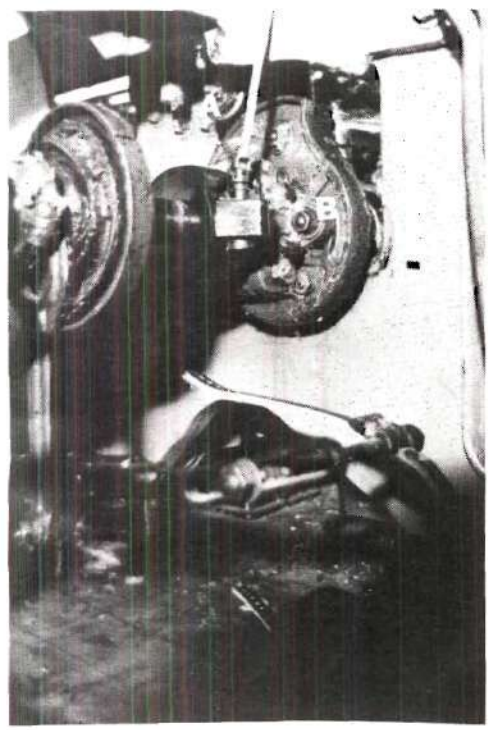


Figure 3. (A) The Pick Ball,
(B) The Pick Cam

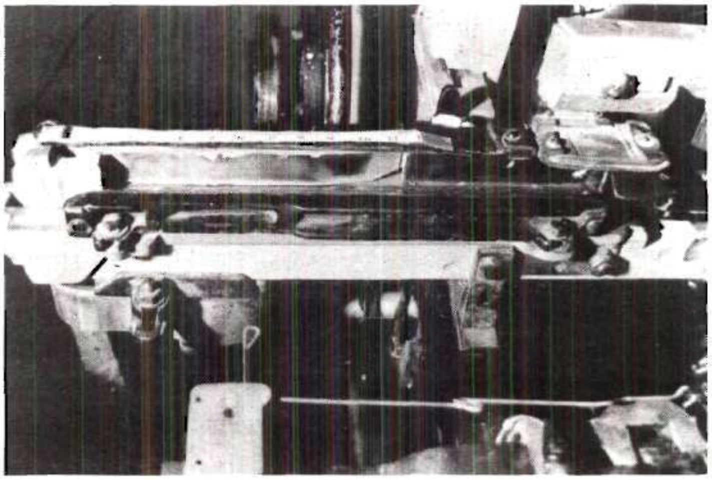


Figure 4. The Left Shuttle Box of the Hunt Automatic Loom Showing Picker (A)

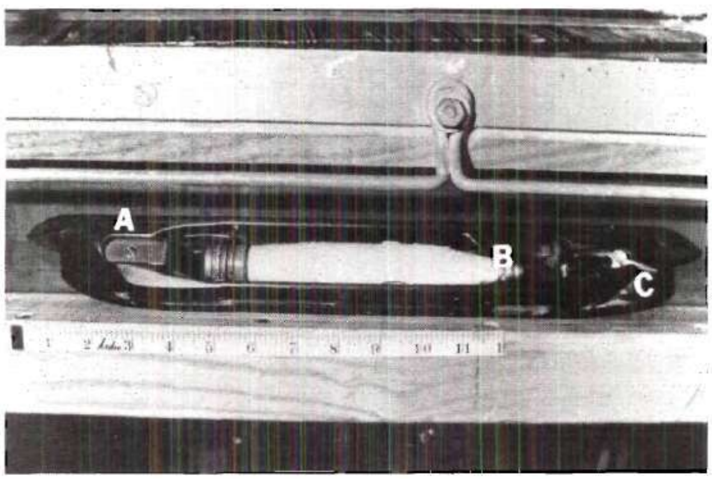


Figure 5. The Fly Shuttle (A) Shuttle, (B) Fixed Bobbin, (C) Eyelet Through Which Weft Passes

wooden levers called picking sticks, one on each side of the loom. The lower end of the stick passes through and is bolted to a curved piece of metal (the rocker) which makes intermittent contact and partly rolls on a piece of metal extending out from the frame (the picking shoe). The picking stick is secured by a spring to this lower component in such a manner that it is normally at rest in the backward position. The picking stick is periodically yanked forward by a cam-actuated strap (this is the basic shuttle throwing action). The strap is not fastened to the stick but is in contact with it during the forward motion. When the cam has revolved sufficiently that this contact is lost, the tendency caused by the spring is for the stick to rock backwards. The point of contact of the strap with the stick is a piece of metal termed the power block which is affixed to the stick. Forward and backward motion of the picking stick is checked by a leather loop at the stick's top. Thus, a typical sequence of motion for a picking stick is as follows: (1) stick in backward position; (2) stick suddenly yanked forward by cam-actuated strap; (3) forward motion stopped by leather loop; (4) stick rotated back because of spring; (5) backward motion stopped by leather loop. Depending on various adjustments, the forward motion of the stick may also be stopped by collision with the end of the slot in the shuttle box within which the top of the stick moves. A small mallet-shaped part (the picker) is attached to the top of the stick and it is this which actually impacts with the shuttle. The catching, or deceleration, of the shuttle is termed boxing while the throwing, or acceleration, of the shuttle is termed picking out.

The cams causing the alternate forward yanking of the picking sticks are both on the same shaft and rotate clockwise as viewed from the left side of the loom. The oscillating roller follower, which actually pulls the strap which yanks the stick, is called the pick shaft. The roller is termed the pick ball.

Other motions which take place during loom operation include the raising and lowering of the harnesses (which raise and lower the warp threads forming the shed). This action is accomplished with cams, which raise and lower plane-face oscillating followers called treadles connected to the spring-loaded harnesses in the base of the loom.

In addition to the motions just described, the sley, boxes, picking sticks assembly is rotated about an axis passing through the lower front of the loom frame (the bottom rocking shaft). The comb-like reed, attached to the sley and separating the warp threads, pushes the weft close to the previously woven cloth where it is locked in place by the harness-driven newly formed shed. This pushing is called the beat-up motion. It is driven by a crank-rocker type four-bar linkage, the reed-carrying assembly being the rocker link. Gears transmit power from the crank link of this four-bar to the other loom shafts. The crankshaft is driven, either through gears or belts, by the loom's motor.

Several protective mechanisms (to minimize downtime) are included in the loom. A weft-sensing fork is pushed into the weft after each pick. If the weft is not in place (due to thread breakage or an empty bobbin), the loom is stopped. A feeler arm is pushed into the bobbin when the shuttle enters the box to determine whether the bobbin is low on thread. If the bobbin is almost empty, it is replaced on the next pick. Should a warp thread fail, a thin conductor hanging on it drops to complete a circuit which causes the loom to stop. Also, should the shuttle get caught in the shed, a mechanical sensing device in the shuttle box determines that the shuttle has not arrived and stops the beat-up motion to keep the warp threads from breaking in large numbers.

The primary noise generating motions are those associated with accelerating and decelerating the shuttle. In some looms (those with gear driven crank shafts) gear noise can also be significant. Surveys have shown that weaving mills, where sometimes hundreds of fly shuttle looms exist under one roof, are among the noisier industrial environments with noise levels on the order of 102 dB(A) [Karplus and Bonvallet, 1953] (well above the government set standards). Hanson (1974) estimates that over 250,000 people are subjected to weave room noise on a daily basis. Such exposure constitutes a hearing damage risk [Burns, Hinchcliffe, and Littler, 1964; Atherly and Noble, 1968; Walz, 1969] and detracts from the quality of life for those affected workers. Almost all natural fiber weaving is done on fly shuttle looms. Noise reducing modifications for existing machines are desirable for this reason.

The purpose of the present work is to estimate the contribution of each vibrating loom component to the overall sound level caused by the machine. In subsequent work, attention may be directed to the greatest noise sources first and to the development of suitable noise reducing modifications for fly shuttle looms.

CHAPTER II

SURVEY OF PREVIOUS AND CONTEMPORARY WORK ON FLY SHUTTLE LOOM NOISE REDUCTION

In early 1965, A. Stott, works manager of the canvas works of Nairn-Williamson Ltd. (GB), reported a 10.5 dB noise reduction in a weaving mill when polyethylene picking points, treadle balls and picking cones, nylon drive pinion and crank bearings were substituted for their metallic counterparts in fly shuttle looms. Actual noise levels were not stated, and no attempt to determine the effect of each separate modification was indicated. Later attempts of a similar nature [Taylor, et al., 1967; Cudworth, 1972, 1973; Hanson, 1974] gave substantially lower reductions than Stott's and it is reasonable to assume that the maintenance operations, performed in the installation of the replacement parts, played a large role in reducing the noise levels.

Taylor et al. (1967) experimentally determined the noise levels associated with loom operation at various stages of assembly. The one-third octave band noise spectra were recorded for a point nine feet from the center of a Blackness loom in a semi-reverberant room before and after gearing, sley, treadles, picking sticks and shuttle were successively added. It was noted that a 12 dB increase over the entire frequency range accompanied the addition of the picking sticks, and that shuttling the loom resulted in an additional 5 dB elevation.

After the installation of polyethylene pickers, picking points, picking bowls and shuttle tips, nylon drive pinion and crank bearings, the experiment was repeated and it was found that the maximum noise reduction attainable through these measures was on the order of 2 dB in the frequency range of .5 KH_z to 2 KH_z. Further experimentation showed that the substitution of polyurethane pickers for polyethylene pickers in an otherwise standard loom resulted in a decrease in sound pressure level of 2 to 3 dB over the frequency range of .8 KH_z to 12 KH_z. Taylor et al. concluded that, while these modifications would not lower fly shuttle loom noise to a safe level, they were a step in the right direction insofar as any reduction of the loom noise was desirable, even if only a few dB.

Although the application of sound absorbing materials to the weave room ceiling, walls, and floor, as outlined by Mills (1969), substantially reduced noise in periphery areas, it had little effect on noise levels in the near vicinity of machines. Mills replaced the metallic pinion gears with nylon pinion gears, fitted the automatic bobbin reject transfer hammer with a plastic head, replaced the standard pickers with durolen plastic pickers, and substituted a felt composition shuttle check for the standard leather shuttle check. These measures resulted in a 2 to 3 dB noise reduction. Mills also experimented with a hydraulic picking stick check, but this latter modification was discarded due to its short life expectancy.

With a technique which involved experimentally recording the sound pressure level as a function of time over a complete loom cycle, Cudworth (1972, 1973) evaluated the effectiveness of the installation of various plastic components in reducing impact noise peaks. This research, sponsored by Draper Division of North American Rockwell (a major manufacturer of fly shuttle looms), is continuing at the Liberty Mutual Research Center in Hopkinton, Massachusetts.

In the Liberty Mutual experiments measurements were made on a Draper X2 fly shuttle loom located in a semi-anechoic chamber. Polyurethane parts were substituted for the standard pickers, power blocks, pick balls and pick shaft bearing blocks with the result that peak instantaneous sound pressure levels decreased by as much as 16 dB. Hanson (1974), also of Liberty Mutual, stated in a recent paper that the first three substitutions resulted in decreased sound pressure peaks for both picking out and boxing. The latter substitution affected picking out sound peaks only. The slow A-weighted response generally dropped from 5 to 7 dB with the estimate that weave room noise could be attenuated by as much as 12 dBA under ideal conditions. Hanson indicated that work is being directed toward determining the field performance of the polyurethane components and that future research will be aimed at kinematic modification of the loom.

At North Carolina State University, Emerson (1973) and his colleagues modified a test loom by replacing the four-bar linkage reed drive with a cam reed drive. Proper design of the cam profile increased the length of time the shed was open. This allowed a lower shuttle velocity which in turn reduced impact forces in the picking out and boxing motions. Emerson suggested that separating the shuttle box, picking stick assembly from the reed would allow further linkage modifications and would also allow the possibility of an enclosure being used to cloak the entire picking operation. This program was apparently recently temporarily discontinued, though it may have resumed. It is our general understanding that textile machinery noise reduction is a long term research activity at North Carolina State.

Based on the premise that noise in the fly shuttle loom is due to vibration caused by the impact forces, associated with the boxing and picking out operations, it would appear that past efforts have concentrated on loom modifications with three purposes in mind:

1. To impede the transfer of energy between components and reduce impact forces through the use of plastics in place of metal [Stott, 1965; Taylor et al., 1967; Mills, 1969; Cudworth, 1972, 1973; Hanson, 1974].

2. To reduce the impact forces through kinematic modification of the fly shuttle loom [Emerson, 1973; Hanson, 1974].

3. To absorb the sound through the use of acoustic tile or enclosures [Mills, 1969; Emerson, 1973].

Each study has involved one's making a modification, measuring the result, and subsequently making another modification.

CHAPTER III

ESTIMATION OF SOUND PRESSURE LEVELS DUE TO VIBRATING MACHINE COMPONENTS

There is general agreement that noise in the fly shuttle loom is due to component vibrations induced by the acceleration and deceleration of the shuttle [Taylor et al., 1967; Cudworth, 1972, 1973; Emerson, 1973; Hanson, 1974]. It can be shown that the sound pressure level due to a vibrating surface, where spherical spreading is assumed, can be estimated by the relation

$$L_p = 119 + 20 \log \frac{a}{f_c} + 10 \log \frac{A}{r^2} \tag{1}$$

subject to the conditions and definitions below. For cylindrical spreading the above becomes

$$L_p = 122 + 20 \log \frac{a}{f_c} + 10 \log \frac{A}{r\ell} \tag{2}$$

For plane radiation, as would correspond to the sound pressure level very near a vibrating panel, the analogous relation is

$$L_p = 130 + 20 \log \frac{a}{f_c} \tag{3}$$

Here, a is the root mean square acceleration in meters per second squared, for a representative point on the surface, for a particular octave band, f_c is the center frequency of the octave band, A is the total exposed area of the surface, ℓ is the length of the surface where applicable, r is the radial distance between the surface and the point at which the sound pressure level is to be estimated, and L_p is the sound pressure level in dB re 2×10^{-5} N/m^2 for the octave band of interest.

The development of the above equations involves several assumptions, the most important being that

$$p = \rho c v \tag{4}$$

at the surface. Here p is the acoustic pressure, ρ is the density of air, c is the speed of sound in air, and v is the fluid velocity, equal to normal component of surface velocity.

This assumption is good for frequencies above the coincidence frequency [Beranek, 1971]. Below the coincidence frequency predicted sound pressure levels may be considerably larger than actual sound pressure levels. Theoretical development of these equations with other pertinent information regarding their applicability is presented in Appendix A.

CHAPTER IV

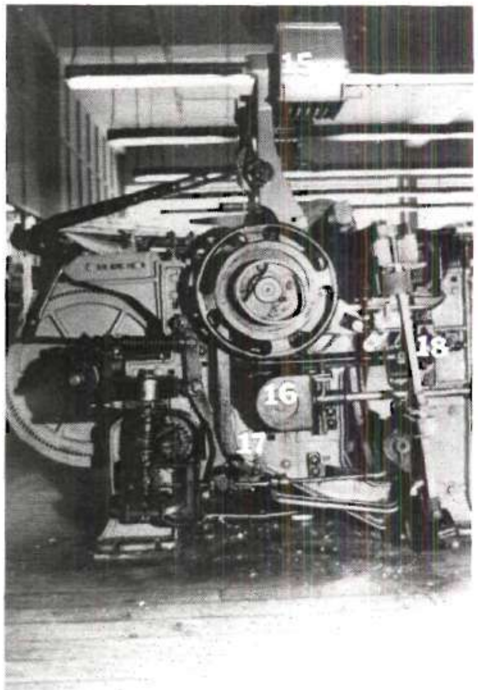
ESTIMATION OF THE SOUND PRESSURE LEVELS DUE TO COMPONENTS OF THE HUNT AUTOMATIC LOOM

Instrumentation and Procedure

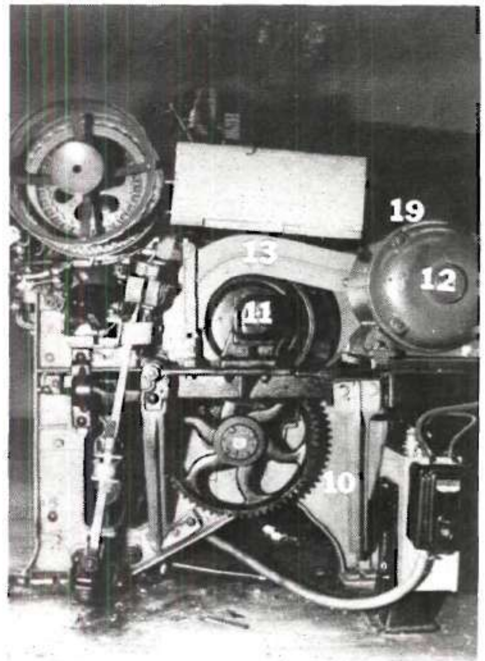
For estimation of the relative contribution of each part of a fly shuttle loom to the overall sound pressure levels caused by the loom, the Hunt Automatic Loom located in the weaving lab of the School of Textile Engineering at Georgia Institute of Technology was divided into the 23 surfaces shown in Figure 6. The octave band acceleration spectrum was determined for a typical point on each surface. Then appropriate surface dimensions and the radius to a common reference point were measured to facilitate the application of the predictive equations given in Chapter III.

In the recording of the octave band root mean square acceleration spectrum of each surface shown in Figure 6, the following instrumentation of Brüel and Kjaer manufacture was used:

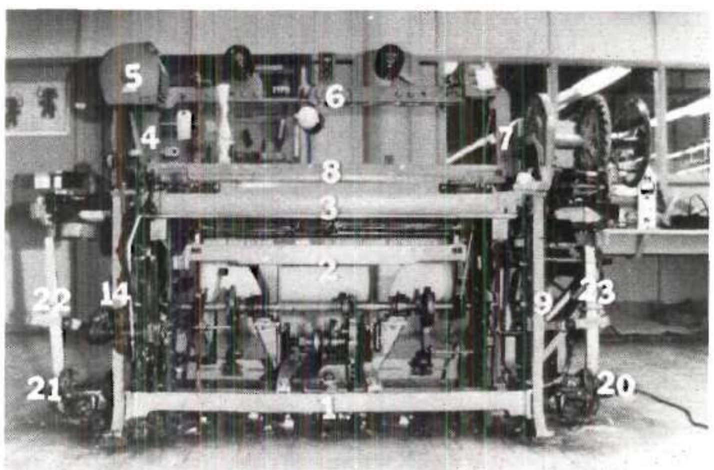
1. Accelerometer type 4336, Serial No. 199240
2. Input stage type ZC0007
3. Inpulse Precision Sound Level Meter type 2204, Serial No. 362502
4. Octave Filter Set type 1613, Serial No. 370455



Left Side



Right Side



Front

Figure 6. The Loom Divided Into 23 Surfaces
(See Table 6)

The accelerometer was press fit into holes drilled in the picking stick surfaces. Wax was used to secure it to the surface at all other points.

The data was taken with the meter switch set on "slow" and the weighting network switch set on "ext. filter." The weighting potentiometer in the external filter set was off. Calibration was checked and adjusted using the Impulse Precision Sound Level Meter's internal reference voltage before measurements were made at each point.

During the experiment, the loom ran at 167 picks per minute corresponding to a crankshaft speed of 167 revolutions per minute and a picking camshaft speed of 83.5 revolutions per minute. The warp and harnesses were removed to eliminate the possibility of shutdown due to thread breakage. The weft sensing fork, automatic bobbin reject transfer mechanism, warp let off mechanism, warp drop wire circuit and treadles were disconnected to allow operation in this condition. Data is presented in Appendix B.

The total exposed area of each surface was measured and a point, one meter from the center of the loom's front face 1/2 meter from the floor, was chosen as the reference point for forming estimates based on accelerometer data of sound pressure levels. Long slender bodies, such as the picking sticks, were considered as cylindrical radiators and their length was recorded. The radial distance from each surface to the reference point was also recorded. (See

Appendix B).

The octave band sound pressure levels in dB re 2 x 10^{-5} N/m^2 were recorded at the reference point during loom operation. The following Brüel and Kjaer instrumentation was used:

1. Microphone type 4145, Serial No. 346747
2. Random Incidence Corrector type UA0055
3. Goose neck type UA0196
4. Input stage type ZC0007
5. Impulse Precision Sound Level Meter type 2204, Serial No. 362502
6. Octave Filter Set type 1613, Serial No. 370455
7. Tripod
8. Pistonphone type 4220, Serial No. 274090

Data was taken while all other machinery in the room was off. The meter switch was set on "slow" and the weighting network switch set on "ext. filter." The weighting potentiometer in the external filter set was off. Calibration was performed with the pistonphone prior to making measurements. Octave band ambient noise levels were recorded; the appropriately corrected fly shuttle loom noise levels at the reference position are presented in Table 1.

Data Analysis and Concluding Remarks

Estimated sound pressure levels due to each surface are given in Table 3. The sources are listed there according to the order of magnitude of their contributions for each octave band.

The summation of the estimated sound pressure levels over all the areas results in the values presented in Table 2. Actual sound pressure levels are also listed there for comparison purposes.

As expected, there is poor agreement at lower frequencies due to the inherent inaccuracy of the assumptions. For the 1000 H_z through 31500 H_z octave bands, predicted levels are within 13 dB of the actual levels. It should be noted that the analytic models on which the estimates are based are designed ideally for application in an anechoic chamber. The Hunt Automatic Loom used for this study is actually located in the corner of a semi-reverberent room. Taking this fact into account would tend to raise the magnitudes of the estimated sound pressure levels, but it was estimated that this correction would be minor.

The models predict the picking sticks to be the loudest noise sources in every octave band. Actual sound pressure levels are greatest in the 500 H_z through 8000 H_z octave bands. In each of these bands the two picking sticks are separated from the third loudest source by 24 dB or more. One may tentatively conclude that the most effective fly shuttle loom modifications should be those which either reduce picking stick vibration levels or absorb the sound radiated by the picking sticks.

Table 1. Actual Sound Pressure Levels at the Reference Position

Octave (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Actual L_p	67	72	77	77	82	86	90	87	82	73	49

Table 2. Comparison of Estimated and Actual Sound Pressure Levels at the Reference Position

Octave (Hz)	31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Estimate L_p	122	116	109	112	103	99	92	86	75	65	50
Actual L_p	67	72	77	77	82	86	90	87	82	73	49

Table 3. Rank Order of Vibrational Noise Sources (First Number Refers to Surface List in Figure 6. Second Number is Estimated L_p at the Reference Point Due to that Surface.)

Octave Center Frequency, Hz										
31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Estimated L_p due to Surface in Rank Order										
23-121	23-114	23-107	23-110	22-100	23-97	23-90	22-83	22-74	22-63	22-47
22-114	22-110	22-103	22-107	23-100	22-95	22-88	23-83	23-70	23-61	23-47
8-103	21-99	20-96	20-90	20-77	9-70	16-63	16-56	16-46	2-31	3-21
21-102	8-94	21-92	21-82	10-74	2-69	9-61	9-53	9-44	21-27	9-18
20-98	20-94	9-87	3-78	3-73	10-69	2-60	17-53	2-38	9-26	21-18
3-95	3-88	3-84	9-77	9-73	20-68	17-59	2-48	17-38	3-24	2-17
7-92	6-86	8-82	17-77	2-72	17-66	21-59	10-48	3-36	16-24	17-14
17-92	9-85	2-80	2-75	21-70	21-64	10-57	3-46	21-33	17-21	1-10
6-89	7-83	17-79	8-75	1-69	1-63	3-55	1-45	1-32	1-15	20-5
5-84	14-82	1-78	10-73	17-69	3-63	20-55	21-39	10-32	10-13	16-2
99-84	17-82	14-77	1-71	16-68	16-57	1-54	13-38	13-24	20-11	10-0
2-83	2-80	10-76	16-71	8-65	8-54	8-44	20-35	5-20	8-5	5--6
14-82	1-79	11-74	4-66	4-62	13-50	13-44	5-30	14-17	5-5	14--8
1-81	16-79	16-74	14-65	7-61	14-47	5-37	8-30	11-16	13-4	8--10
4-81	4-77	6-70	7-64	13-56	4-45	14-36	14-27	20-16	14--2	13--10
19-77	5-76	7-70	6-62	14-54	7-42	15-33	15-22	8-15	4--4	4--16
10-72	10-74	15-70	11-62	5-53	5-40	4-31	4-20	6-10	15-7	7--16
12-71	18-71	18-69	13-61	6-53	11-40	7-28	11-19	15-9	7--10	6--17
15-71	11-69	4-66	18-59	11-52	6-39	18-27	7-16	4-8	6--11	18--23
13-70	15-69	5-59	5-27	15-45	15-38	6-26	6-13	18-3	11--12	11--24
18-67	13-66	13-57	15-52	18-45	18-38	11-25	18-13	7-1	18--14	15--25
11-65	19-60	12-45	12-33	19-26	12-25	12-11	12-0	12--5	12--29	19--32
16-64	12-56	19-45	19-33	12-25	19-12	19-6	19--1	19--16	19--30	12--33

CHAPTER V

RECOMMENDATIONS FOR FUTURE FLY SHUTTLE LOOM NOISE RESEARCH

The picking stick (Figure 2) is essentially a wooden beam used as a lever to catch and throw the shuttle during weaving. The lowest natural frequency of a picking stick, considered as a "hinged-free" beam [Den Hartog, 1956] is on the order of 50 H_z. The vibrational frequencies of interest (500-8000 Hz) are well above 50 H_z and consequently many modes of vibration are excited. For this reason, making the picking stick stiffer (by modification of the cross section or use of a material other than wood) is unlikely to be a significant means of reducing vibration.

Maintenance requirements make the enclosure of the picking stick impractical and economic factors make kinematic modification of existing machines to decrease impact forces undesirable. The substitution of parts in the picking mechanism by alternate parts made out of polyurethane will reduce the stick's vibration levels. However, one would not expect to achieve the maximum noise reduction by this method.

Two methods of noise control, either of which, if proved feasible, would lead to an inexpensive solution to the loom noise problem may be suggested. The first, and possibly the most promising, is that of attenuating the radiated sound by covering the picking sticks with a porous blanket wrapped with an impermeable membrane ("lagging," Figure 7). A suitable lagging treatment could probably be selected on the basis of the following considerations:

1. Insertion loss for frequencies within the 500 H_z to 8000 H_z octave bands.
2. Durability and field life expectancy.
3. Ease of application.
4. Cost.

This type of solution would offer the noise reduction benefits of an enclosure, without the penalty of maintenance difficulties.

A second possibility is the application of a surface damping treatment to reduce picking stick vibration within the same frequency ranges (Figure 8). This would require a greater design effort than required for the lagging design, but the net result could conceivably be extremely simple to implement.

It is recommended that short term research be directed toward the development of practical "retrofit" solutions such as described in the preceding paragraphs. However, in view of the fact that the general mechanisms of noise production by fly shuttle looms are still not well understood, and with the premise that the best solution would

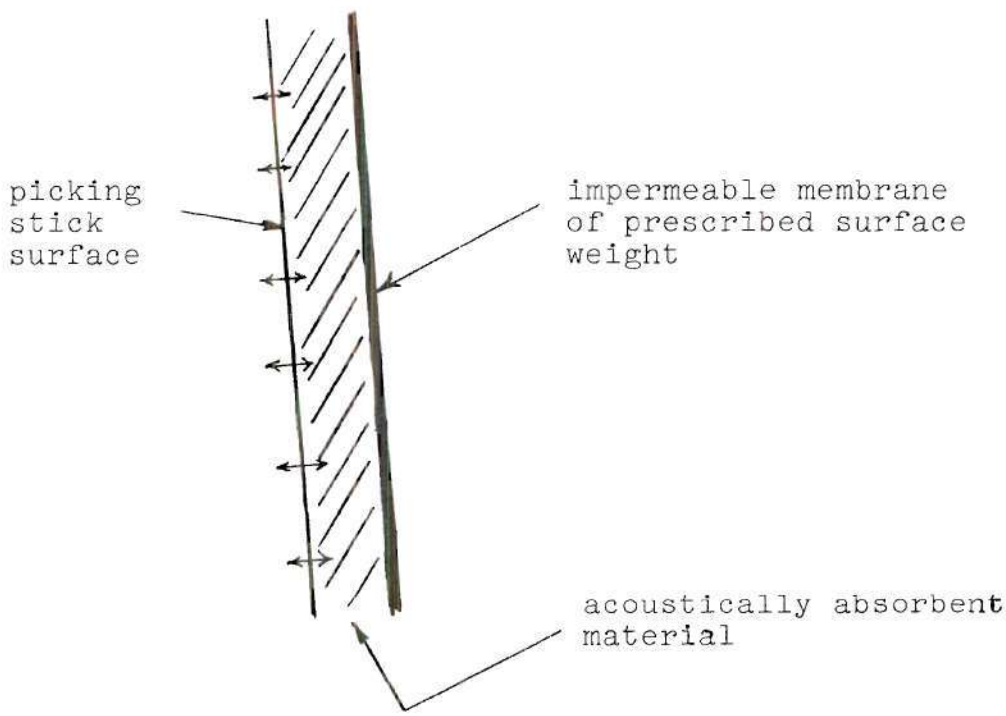


Figure 7. Lagging the Picking Stick

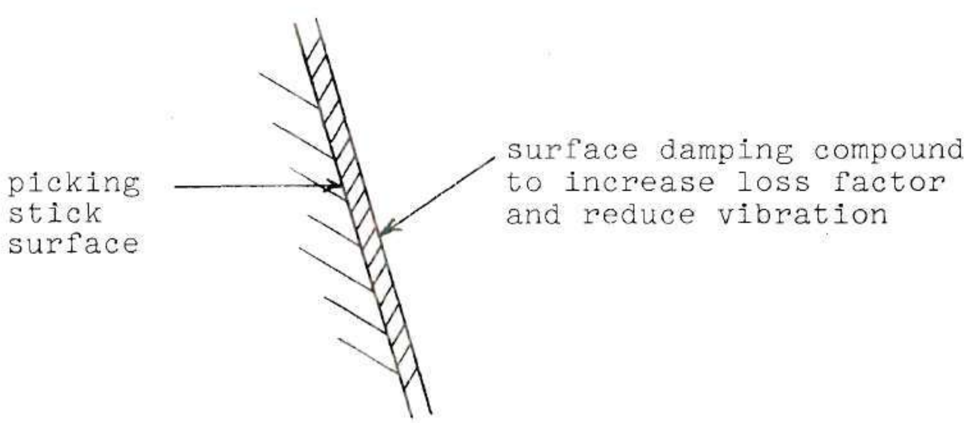


Figure 8. Surface Damping the Picking Stick

require a fairly detailed understanding of such mechanisms, it is recommended that long term research also be carried out. Such research should be of a general nature and should involve the application of the basic principles of physics to assure applicability of results to the broadest possible spectrum of industrial noise problems.

APPENDICES

APPENDIX A

SOUND PRESSURE LEVEL DUE TO A VIBRATING SURFACE

Consider a vibrating surface radiating sound into space as shown in Figure 9. With the assumption that at any point on the surface the acoustic radiation behaves as a plane wave, one has

$$p = \rho c v \tag{1}$$

where p is the acoustic pressure just outside the surface, ρ is the density of air, c is the speed of sound in air and v is the fluid velocity just outside the surface which is equal to the normal velocity of the surface.

This assumption will be good [Beranek, 1971] for frequencies above the coincidence frequency (i.e., the frequency at which the phase velocity of waves traveling in the medium is equal to the speed of sound in air). For transverse vibrations of the picking sticks, this coincidence frequency is on the order of 350 Hz; for steel plates it is roughly equal to 500/h, where h is the thickness of the plate in inches.

The power radiated per unit area is pv, hence the time average of the total power is

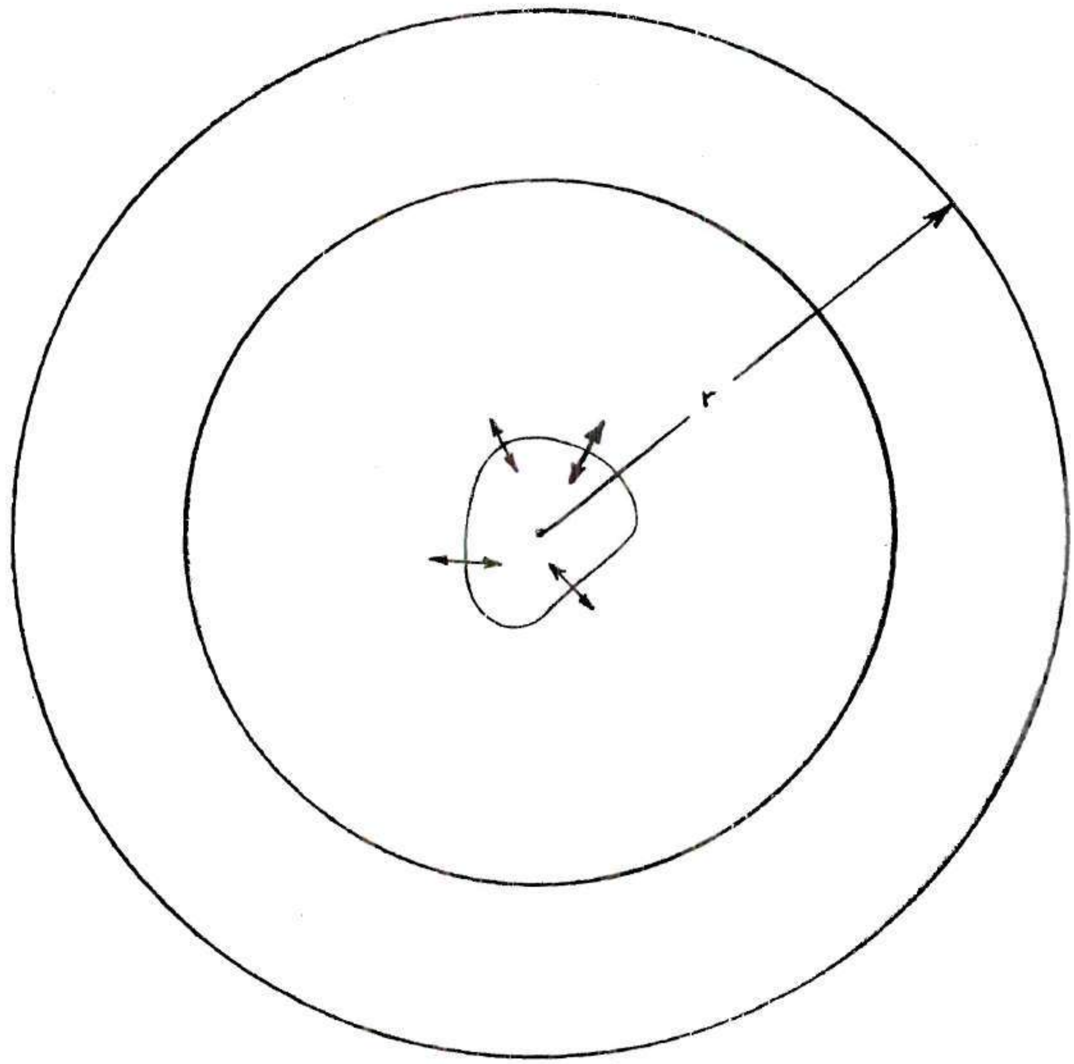


Figure 9. Vibrating Surface Radiating Sound Into Space

$$W = \int_{\text{surface}} \rho c \, \langle v^2 \rangle \, dA \tag{2}$$

where $\langle x \rangle$ indicates the time average of x. We will approximate this integral by

$$W = \rho c \, \langle v^2 \rangle_{rep} A \tag{3}$$

where W is the acoustic power output of the surface, A is the total exposed surface area, and $\langle x \rangle_{rep}$ is the value of $\langle x \rangle$ at a representative point on the surface.

For spherical spreading the intensity at a distance r from the source is given by

$$I = \frac{W}{4\pi r^2} \tag{4}$$

If r is sufficiently large, the wavefront will be close to that of a free traveling plane wave. If p and v are in phase we may write

$$I = \frac{\langle p^2 \rangle}{\rho c} \tag{5}$$

It follows that, in any frequency band b,

$$\langle p^2 \rangle_b = \frac{(\rho c)^2 A \langle v^2 \rangle_{rep,b}}{4\pi r^2} \tag{6}$$

Then, with the approximation that

$$\langle v^2\rangle_{rep,b} = \frac{\langle a^2\rangle_{rep,b}}{(2\pi f_c)^2} \tag{7}$$

where a is the normal acceleration of the surface, and where f_c is the center frequency of the band, we may write an expression relating the normal acceleration of the surface in a given frequency band to the sound pressure level in that same band, i.e.

$$L_p = 10 \log \frac{(\rho c)^2 A\langle a^2\rangle_{rep,b}}{16\pi^3 r^2 f_c^{\,2} p_o^{\,2}} \tag{8}$$

where L_p is the sound pressure level in dB re p_o and p_o is the reference pressure. In MKS units this becomes

$$L_p = 119 + 20 \log \frac{a_{rms}}{f_c} + 10 \log \frac{A}{r^2} \tag{9}$$

If cylindrical spreading is assumed, then the intensity would be expressed as

$$I = \frac{W}{2\pi r \ell} \tag{10}$$

where ℓ is the length of the cylinder. An approximate derivation similar to the above leads to the expression

$$L_p = 122 + 20 \log \frac{a_{rms}}{f_c} + 10 \log \frac{A}{r\ell} \tag{11}$$

where all the appropriate quantities are to be taken with MKS units.

Finally, if it is assumed that plane waves are produced by, say, a vibrating plate, one sets

$$p = \rho c v \tag{12}$$

(This should be good above the coincidence frequency.) Then for plane radiation, one has the following expression

$$L_p = 130 + 20 \log \frac{a_{rms}}{f_c} \tag{13}$$

where a_{rms} is the root mean square normal acceleration at the surface.

Note that, in each of the expressions,given for L_p, if $\langle p^2 \rangle$ is off by a factor of G, then

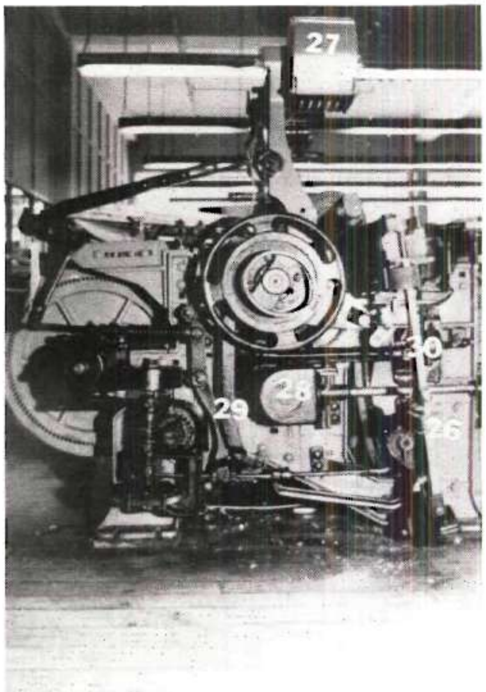
$$|Lp_{actual} - Lp_{estimate}| = 10 \log G \tag{14}$$

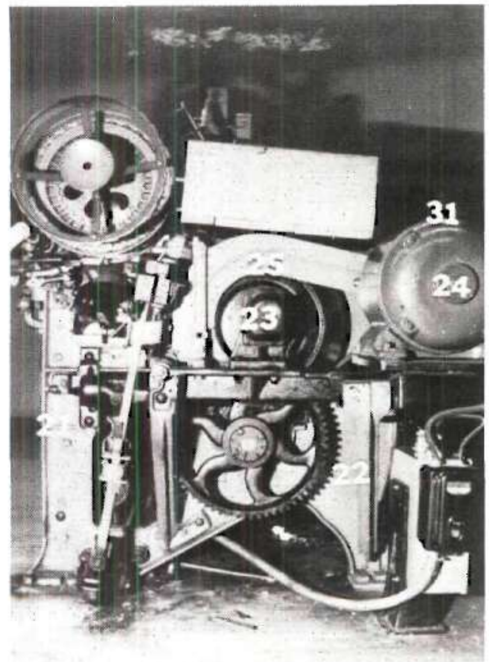
Thus a 10 dB difference in levels corresponds to G = 10, etc.

APPENDIX B

PRESENTATION OF DATA

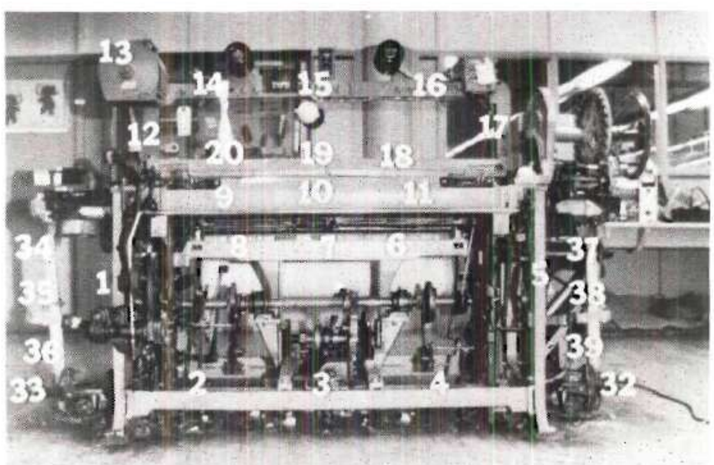
The octave band normal surface acceleration spectrum was recorded for the 39 points shown in Figure 10. These accelerations are presented in Table 4. The loom was divided into the 23 surfaces shown in Figure 6 of the text. The acceleration of a representative point on each surface was taken as the average value of the acceleration recorded at the data points falling on that surface. These representative accelerations are presented in Table 5. Table 6 contains a verbal description of each surface, indicates whether spherical or cylindrical spreading was assumed, and records the value $10 \log \frac{A}{r^2}$ or $10 \log \frac{A}{r\ell}$, for the surface and the reference point, as appropriate.





Left Side

Right Side



Front

Figure 10. The 39 Points for Which Acceleration Spectra Were Recorded

Table 4. RMS Surface Accelerations in MKS Units for the Positions Shown in Figure 10

	Octave center frequency, Hz										
	31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Position	Acceleration in Meters per Second Squared										
1	.58	.47	.36	.19	.22	.18	.19	.14	.08*	.02*	.02*
2	.50	.55	.95	.60	.46	.27	.15	.08	.03	.01	.01
3	.68	1.1	1.4	1.3	2.0	2.6	2.1	1.5	.70	.19	.22
4	.46	.90	2.4	2.3	4.0	3.6	2.6	1.7	.80	.26	.25
5	.65	.76	1.3	1.3	3.0	5.4	3.4	2.1	1.6	.45*	.40*
6	.80	1.2	2.0	3.0	4.3	6.5	3.1	2.3	.90	.83*	.42*
7	.82	1.3	3.2	2.1	3.6	4.2	5.4	2.1	1.4	1.2*	.50*
8	1.1	1.3	2.5	3.2	4.5	5.4	3.0	1.4	1.3	1.3*	.50*
9	4.4	3.6	4.2	4.0	4.5	2.5	1.9	1.3	.80	.40*	.65*
10	2.2	2.2	2.7	3.0	3.1	2.8	1.9	1.3	1.0	.46*	.60*
11	3.2	2.3	3.5	4.2	4.1	2.2	2.0	1.5	.80	.43*	.65*
12	.70	.90	.50	1.0	1.2	.35	.14	.08	.04	.02	.01
13	3.1	2.5	.70	1.1	1.4	.60	.90	.80	.50	.18	.10
14	2.0	2.8	1.2	.60	.45	.20	.10	.05	.02	.01	.01
15	1.9	3.2	.60	.80	.60	.20	.10	.04	.02	.01	.01
16	1.7	2.2	1.0	.60	.40	.19	.07	.04	.01	.01	.01
17	2.5	1.8	.80	.80	1.1	.24	.10	.05	.02	.01	.01
18	9.5	6.7	3.4	3.2	1.7	1.1	.70	.26	.10	.06*	.02*
19	8.5	6.2	2.9	2.3	1.6	.80	.40	.20	.09	.04*	.02*
20	7.1	5.3	3.1	3.1	2.0	1.2	.70	.25	.09	.07*	.02*
21	1.0	2.7	7.7	4.4	4.4	5.0	3.7	3.6	2.5	.50	.35*
22	.40	1.0	2.5	3.7	7.6	8.6	4.6	3.1	1.0*	.22	.10*
23	.56	1.7	6.1	3.2	2.0	1.0	.35	.34	.50	.04*	.02*
24	.80	.28	.15	.08	.06	.13	.05	.03	.01	.004	.01
25	1.5	2.1	1.4	4.5	4.9	7.9	5.2	5.0*	2.1*	.40*	.15*

Table 4 (concluded)

	Octave center frequency, Hz										
	31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Position	Acceleration in Meters per Second Squared										
26	.70	2.2	2.5	1.2	.61	.55	.23	.15	.09*	.02*	.02*
27	.65	.95	2.1	.53	.51	.44	.49	.30	.13	.04	.01
28	.55	6.0	6.7	9.5	14	7.8	32	27	18	3.0	.45*
29	4.0	2.4	3.4	5.3	4.6	6.0	5.6	5.8	2.0	.60	.50
30	.60	2.1	3.0	2.0	.80	.70	.40*	.16	.10*	.03*	.02*
31	1.7	.53	.18	.09	.08	.03	.03	.03	.01	.004	.01
32	20	26	60	65	29	20	9.0	1.7	.42*	.45*	.43*
33	33	47	38	26	12*	12*	14*	3*	3*	3*	2*
34	22	17	20	44	70	120	80	80	100	40	15
35	70	100	70	350	220	210	200	250	80	80	20
36	28	35	50	55	110	90	90	85	120	50	15
37	55	65	75	80	85	100	90	60*	50*	25*	15*
38	150	100	70	450	240	390	350	300	100	90	30
39	65	90	70	65	70	70	50	45*	45*	25	8*

*Input amplifier overload could not be removed by adjusting the amplifier attenuators. Actual acceleration may be slightly larger than the recorded value. Since accelerations are recorded in units of length per time squared, this fact will not greatly affect the predicted L_p; e.g. if the acceleration is off by a factor of two, L_p will only be off by 3 dB.

Table 5. RMS Surface Accelerations in MKS Units for a Representative Point on Each Surface

	Octave center frequency, Hz										
	31.5	63	125	250	500	1000	2000	4000	8000	16000	31500
Surface	Acceleration in Meters per Second Squared										
1	.55	.85	1.6	1.4	2.2	2.2	1.6	1.1	.51	.15	.16
2	.91	1.3	2.6	2.8	4.1	5.4	3.8	1.9	1.2	1.1	.47
3	3.3	2.7	3.5	3.7	3.9	2.5	1.9	1.4	.87	.43	.63
4	.70	.90	.50	1.0	1.2	.35	.14	.08	.04	.02	.01
5	3.1	2.5	.70	1.1	1.4	.60	.90	.80	.50	.18	.10
6	1.9	2.7	.93	.67	.48	.20	.09	.04	.06	.01	.01
7	2.5	1.8	.80	.80	1.1	.24	.10	.05	.02	.01	.01
8	8.4	6.1	3.1	2.9	1.8	1.0	.60	.24	.09	.06	.02
9	.82	1.7	4.5	2.9	3.7	5.2	3.6	2.9	2.1	.48	.38
10	.40	1.0	2.5	3.7	7.6	8.6	4.6	3.1	1.0	.22	.10
11	.56	1.7	6.1	3.2	2.0	1.0	.35	.34	.50	.04	.02
12	.80	.28	.15	.08	.06	.13	.05	.03	.01	.004	.005
13	1.5	2.1	1.4	4.5	4.9	7.9	5.2	5.0	2.1	.40	.15
14	.64	1.3	1.4	.70	.42	.37	.21	.15	.09	.02	.02
15	.65	.95	2.1	.53	.51	.44	.49	.30	.13	.04	.01
16	.55	6.0	6.7	9.5	14	7.8	32	27	18	3.0	.45
17	4.0	2.4	3.4	5.3	4.6	6.0	5.6	5.8	2.0	.60	.50
18	.60	2.1	3.0	2.0	.80	.70	.40	.16	.10	.03	.02
19	1.7	.53	.18	.09	.08	.03	.03	.03	.01	.004	.006
20	20	26	60	65	29	20	9	1.7	.42	.45	.43
21	33	47	38	26	12	12	14	3	3	3	2
22	40	51	47	150	133	140	123	138	100	57	17
23	90	85	72	198	132	187	163	135	65	47	18

Table 6. The 23 Surfaces of Figure 6

Surface	Description	Cylindrical Spreading	Spherical Spreading	$10 \log \frac{A}{r\ell}$ or $10 \log \frac{A}{r^2}$
1	Lower Front Crossmember	X		-6
2	Middle Inner Crossmember	X		-8
3	Breast Beam	X		-7
4	Left Harness Support Post	X		-8
5	Harness Return Spring Drum (Ends)		X	-15
6	Harness Support Member	X		-9
7	Right Harness Support Post	X		-8
8	Reed, Sley	X		-8
9	Right Front Frame Post	X		-6
10	Right Side Frame		X	-9
11	Right Drive Gear Bearing House		X	-19
12	Motor (Sides)		X	-16
13	Drive Belt Guard		X	-23
14	Left Front Frame Post	X		-6
15	Harness Return Spring Drum (Circumference)		X	-14
16	Cloth Takeup Drive Gear Housing		X	-20
17	Left Side Frame		X	-9
18	Pick Shaft Bearing Block		X	-18
19	Motor (Circumference)		X	-17
20	Right Rocker		X	-17
21	Left Rocker		X	-17
22	Left Picking Stick	X		-10
23	Right Picking Stick	X		-10

APPENDIX C

ESTIMATION OF THE SOUND PRESSURE LEVEL DUE TO A PARTICULAR SURFACE

We consider the left picking stick as a particular example. The octave band acceleration spectrum was recorded. at 3 points on the surface and the average normal surface acceleration was then calculated for each octave band.

The geometry of the picking stick strongly suggested that one consider it as a cylindrical source, hence we used the equation

$$L_p = 122 + 20 \log \frac{a}{f_c} + 10 \log \frac{A}{r\ell}$$

to estimate the sound pressure level at the reference position. The picking stick's exposed area was measured, as was its length and the radial distance to the reference position, one meter from the center of the front face of the loom 1/2 meter from the floor. It was found that the exposed area A was 180 in^2, while the length was 30 in. and the radius was 65 in. Hence, one arrived at the value

$$10 \log \frac{A}{r\ell} = -10$$

The sound pressure level in each octave band was then estimated using the formula

$$L_p = 112 + 20 \log \frac{a}{f_c}$$

The results are listed in Table 7.

Table 7. Calculation of Sound Pressure Levels Caused by the Left Picking Stick

Octave Hz	Acceleration from Table 6	$20 \log \frac{a}{f_c}$	Estimated L_p
31.5	40	2	114
63	51	-2	110
125	47	-9	103
250	150	-5	107
500	133	-12	100
1000	140	-17	95
2000	123	-24	88
4000	138	-29	83
8000	100	-38	74
16000	57	-49	63
31500	17	-65	47

APPENDIX D

THE HUNT AUTOMATIC LOOM

Manufacturer: Hunt Machine Works, Greenville, S. C.

Frame No.: C73-58-BR

Motor: Westinghouse Life-Line Type CSP

1 horsepower

3 phase

60 cycle

220 volt

3.5 amps per line

850 rpm loom design

Serial No. 12N8785

Crankshaft Drive: Triple V-belt

Brake: 6V Warner Electric Brake and Clutch

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